

ACTIVE NARROW-BAND VIBRATION ISOLATION OF LARGE ENGINEERING STRUCTURES

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ABSTRACT

We present a narrow-band tracking control method using a variant of the Least Mean Squares (LMS) algorithm [1] to isolate slowly changing periodic disturbances from engineering structures. The advantage of the algorithm is that it has a simple architecture and is relatively easy to implement while it can isolate disturbances on the order of 40-50 dB over decades of frequency band. We also present the results of an experiment conducted on a flexible truss structure. The average disturbance rejection achieved is 40 dB over the frequency band of 5 Hz to .50 HZ.

1. INTRODUCTION

Many engineering structures contain secondary structures that generate periodic steady state disturbances. These include large ventilation fans, or other rotating machinery typically found in tall buildings, generators in dams, escalators or elevators in public buildings and large electric transformers in power stations and commercial buildings to mention a few. The frequency and magnitude of the periodic disturbances may also vary slowly with time. These disturbances being magnified by resonant base dynamics can cause unacceptable levels of vibration in the structure. The propagation of the disturbances into the structure can be very annoying and in many cases can cause fatigue damage or interference with the operation of other sensitive instruments housed within the structure.

To attack this problem, a variety of techniques have been proposed or are being used in practice. The most popularly practiced method is to place a passive mount (passive isolation) between the disturbance source and the structure. A passive mount is essentially a soft spring and damper combination that can significantly reduce disturbance propagation from the source. While a soft passive mount may be necessary to attenuate a broad band disturbance, it introduces a soft connection that is normally unacceptable from structural integrity considerations. A good example from the automotive industry is that a performance car can not have a soft suspension which could increase riders' comfort by reducing propagation of engine and road disturbances.

In the last 15 years, many investigators including those in reference [3-4] have proposed to augment passive mounts with closed loop feedback control systems (active isolation). Such broad band active isolators suffer from the same problems

as passive isolation. Any broad band force feedback control essentially further softens [2,5] the existing passive mount and in-turn lowers the corner frequency to introduce additional isolation. Also, since such control scheme requires gradual gain roll off for stability, it is difficult to obtain good performance over a large frequency band.

We present a modified narrow-band tracking type feedback control method for isolating periodic disturbances by softening the passive mount selectively only at the disturbance frequencies. As a result, general softening of the mount is avoided and structural integrity is not compromised. The controller is capable of tracking disturbance frequencies over a decade and hence can accommodate normal operational variations.

2. EXPERIMENT HARDWARE

The experiment hardware consists of three components: structure, isolator and disturbance source. Figure 1 shows the system configuration and the three components interconnected along an axis perpendicular to the direction of gravity.

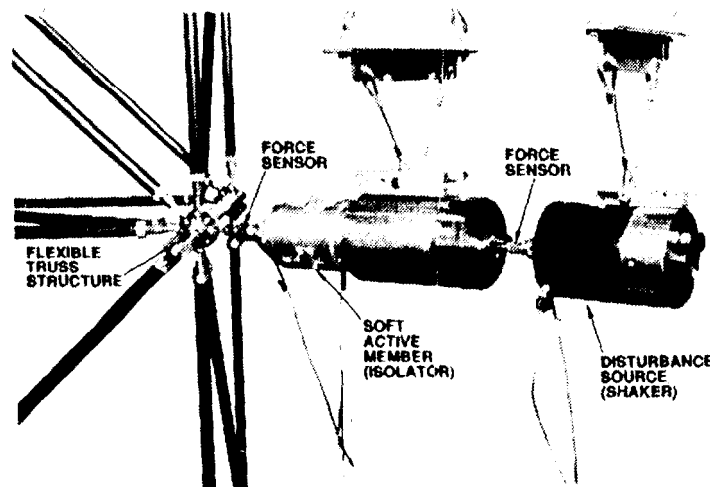


Figure 1: Vibration Isolation Experiment Setup

The structure is an aluminum truss that simulates an elastic structure with a vibrating machinery mounted on it. System identification experiments (figure 3) revealed the presence of approximately 11 lightly damped modes below 50 Hz. The key elements of the isolator are voice coil, a set of flexures and a 4 Kg mass. The stiffness of the flexure along with the mass determines the passive isolation break frequency which in our experiment was about 21 Hz. A load cell (performance sensor) installed between the isolator and the structure measures the force transmitted to the structure and provided the feedback signal in the active isolation control loop. The disturbance source is a proof-mass shaker suspended from the ceiling and attached to the isolation fixture via a stinger connector. The proof mass used in the experiment was approximately 2 Kg. A second load cell (disturbance sensor) measures the force acting on the isolator from the shaker.

3. CONTROL DESIGN

The isolation control system was designed to soften the passive mount at the frequency of a single tone disturbance to improve isolation performance beyond that obtained from the passive stage. The design was carried out in two stages: i) measurement and identification of the plant frequency response function, and ii) design and implementation of the controller. The frequency response function from the voice coil actuator to the force sensor was measured using a Tek2630 Fourier Analyzer and is shown in Figure 3. There are 11 lightly damped modes present between 2 Hz and 50 Hz. Of these, the modes at 5 Hz and 21 Hz are present respectively due to the suspension assembly and the passive flexure mount. The remaining 9 modes are associated with the truss structural. Identification of the plant transfer function by minimizing l_2 norm of the error [9] revealed that the modal clappings for all the structural modes are less than 0.5%. It is interesting to note that the structural modes above the passive break frequency, as predicted by [3], are decoupled. The degree of decoupling of each of these modes depends on the separation between the frequency of the mode and the passive break frequency.

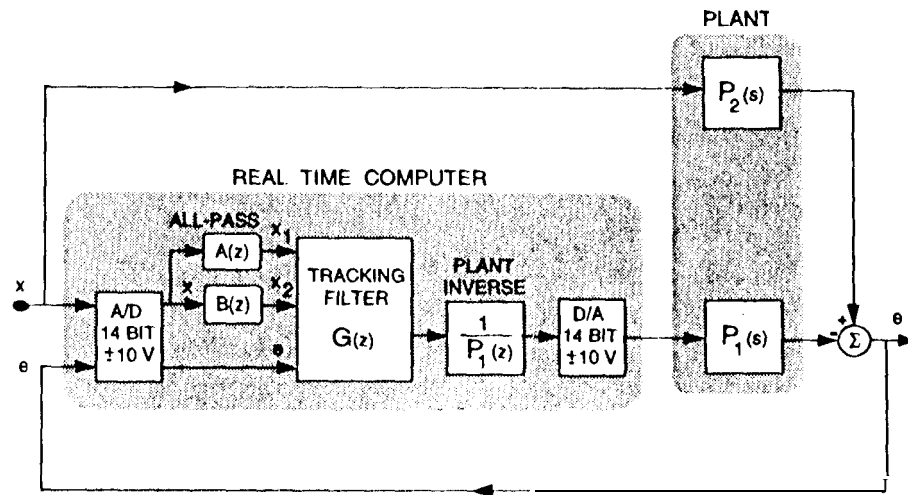


Figure 2: Control System Architecture

The control system consisted of two parts implemented in series: 1) an approximate plant inverse filter and 2) a modified LMS filter. The 9th order plant inverse filter combined with the plant kept the overall phase variation within $\pm 60^\circ$ from 3 Hz to 75 Hz. This was necessary to maintain an overall phase margin of at least 30° for stability of the closed loop system since the LMS filter tracked the frequency varying reference input between 5 Hz and 50 Hz. The combined plant inverse filter and the plant response was measured and is shown in Figure 4. The phase variation is $\pm 55^\circ$ that guaranteed a net phase margin of 35° for stability.

The LMS filter is a tracking type filter that converges to a notch [1,7] of the form given below at the frequency of the reference input which is correlated with the disturbance being isolated.

$$2C^2\mu \frac{s}{s^2 + \omega_0^2} \quad (1)$$

In expression (1), the parameters C and ω_0 are respectively the amplitude and frequency of the reference input and μ is the LMS convergence parameter. Figure 2 shows a schematic of the filter. The transfer functions $A(z)$ and $B(z)$ are 3rd order all pass filters [6,7] that generate two signals 90° apart in phase over a decade of frequency of 5 Hz to 50 Hz. The s -domain equivalents of these two filters is shown below

$$A(s) = \frac{(s - 1036.03)(s - 140.837)(s - 32.655)}{(s + 1036.03)(s + 140.837)(s + 32.655)} \quad (2)$$

$$B(s) = \frac{(s - 302.26)(s - 70.059)(s - 9.527)}{(s + 302.26)(s + 70.059)(s + 9.527)} \quad (3)$$

The response of these two filters was measured and is given in Figure 5. Note that the phase difference of the two outputs did not depart by more than 0.6° from the desired value of 90° . These two signals along with the error signal (transmitted force) were inputs to the LMS algorithm. The combined compensator, discretized at 4095 Hz, was implemented on a Heurikon H KV4F/33MHz 68040 processor running under the VxWorks operating system.

4. EXPERIMENT

We carried out four sets of experiments to demonstrate the performance of the proposed active isolation scheme. In the first and second experiments, the disturbance shaker was excited at 5 Hz and 50 Hz respectively and the transmitted forces to the structure with and without the control loop closed were measured. The disturbance transmission at 5 Hz and 50 Hz were attenuated by 45 dB and 30 dB respectively as shown in Figures 6 and 7. In the third experiment the input disturbance was tuned to 30.75 Hz --a resonance frequency of the structure. The disturbance transmission was reduced by 38 dB by closing the control loop (Figure 8). In the last and the most conclusive experiment, the input disturbance frequency was varied slowly from 5 Hz to 50 Hz. Figure 9 shows the open loop and the closed loop disturbance transmissibilities measured by taking the ratio of the readings of the performance load cell to that of the disturbance load cell at each

frequency point. The reduction in the force transmissibility is over 40 dB from 5 Hz to 25 Hz and over 30 dB to 50 Hz.

5. CONCLUSION

A narrow-band tracking controller was designed and implemented to demonstrate active vibration isolation of steady state harmonic disturbances from a structure over a decade of frequency. Experimental results show that the disturbance isolation is in the order of 40 dB over the frequency range of 5 Hz to 50 Hz. Our implementation required designing an approximate plant inversion filter over the same frequency range. As a result the robustness of the method becomes dependent upon two conditions: i) the plant dynamics does not change appreciably over time and ii) sufficient modal damping is present to accommodate small parameter variations. Though the structure has light damping, its modal properties during the experiment did not change appreciably that could upset the stability condition, Since we can not expect this kind of invariability of modal properties from an engineering structure, the presence of a few percentages of modal damping will increase the robustness of the proposed controller against plant change.

In many cases the structure of the disturbance source maybe large and it is not practical to put an active isolator and force sensor in series and embedded between the primary structure and the disturbance source. A practical approach will be to install a proof mass actuator in parallel and close to the passive mount. The force measurement can be accomplished by installing strain gages directly on the mount. One important difference between the implemented configuration and the suggested configuration is that the latter does not form a collocated actuator and sensor set, As a result, controller design for the latter case is usually more difficult and it is currently under investigation.

Our future research will focus on a multi-axis vibration isolation stage, The stage is equipped with six passive and active mounts. The interaction of MIMO loop control on the multi-axis isolation stage will be of particular interest.

6. ACKNOWLEDGMENT

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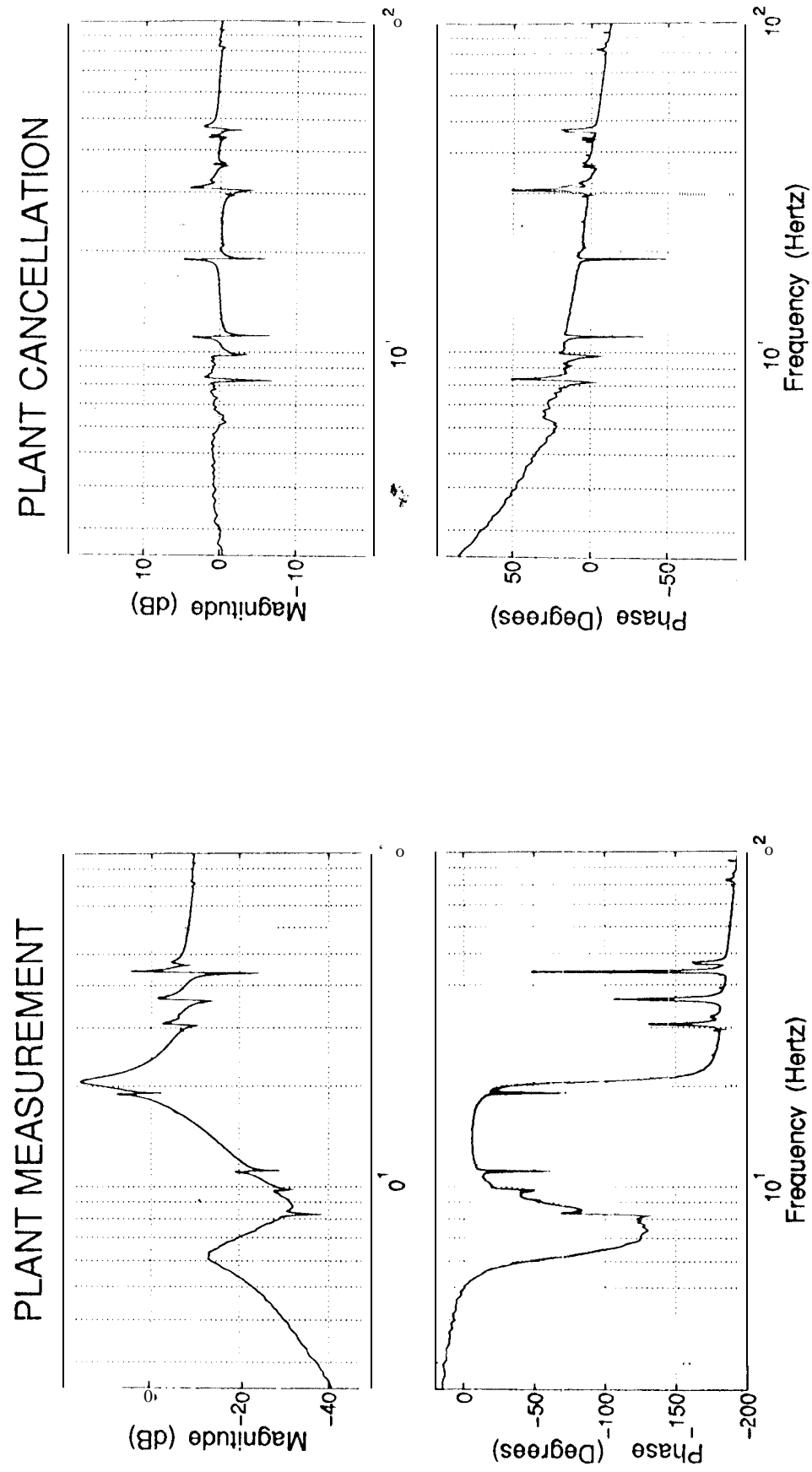


Figure 3: Plant frequency response function

Figure 4: Plant Cancellation (measured)

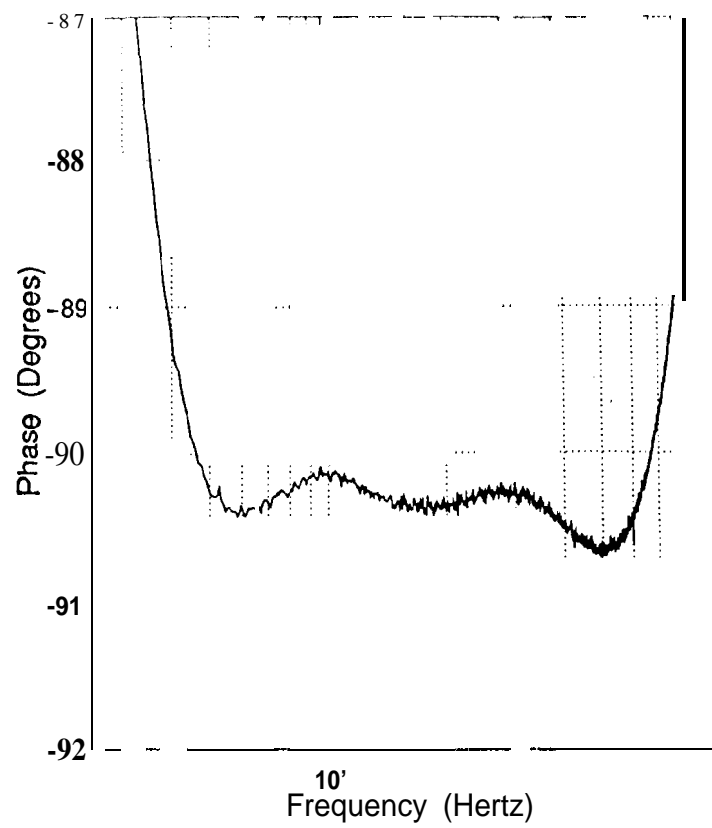


Figure 5: Phase Quadrature (measured)

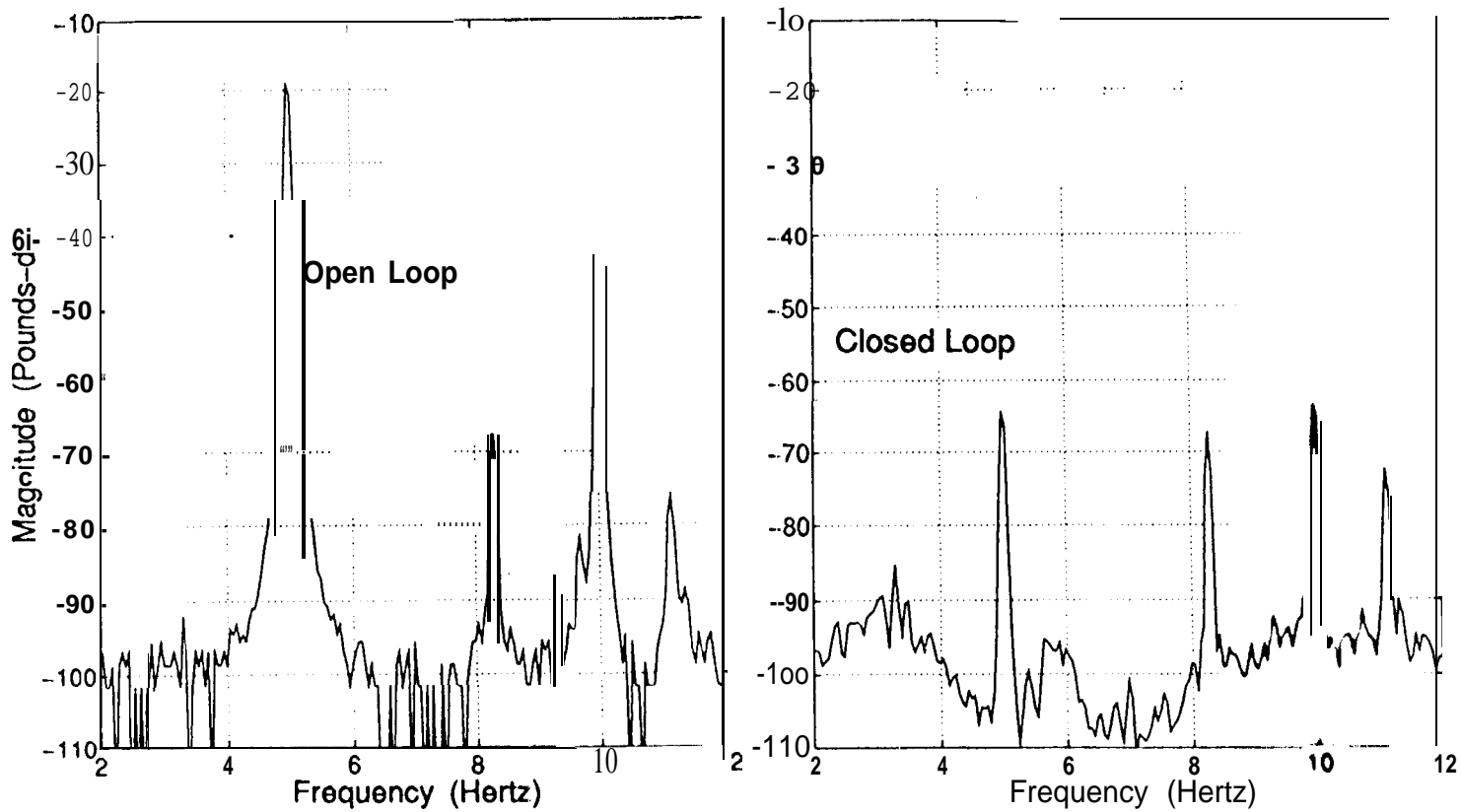


Figure 6: Isolation at 5 Hz

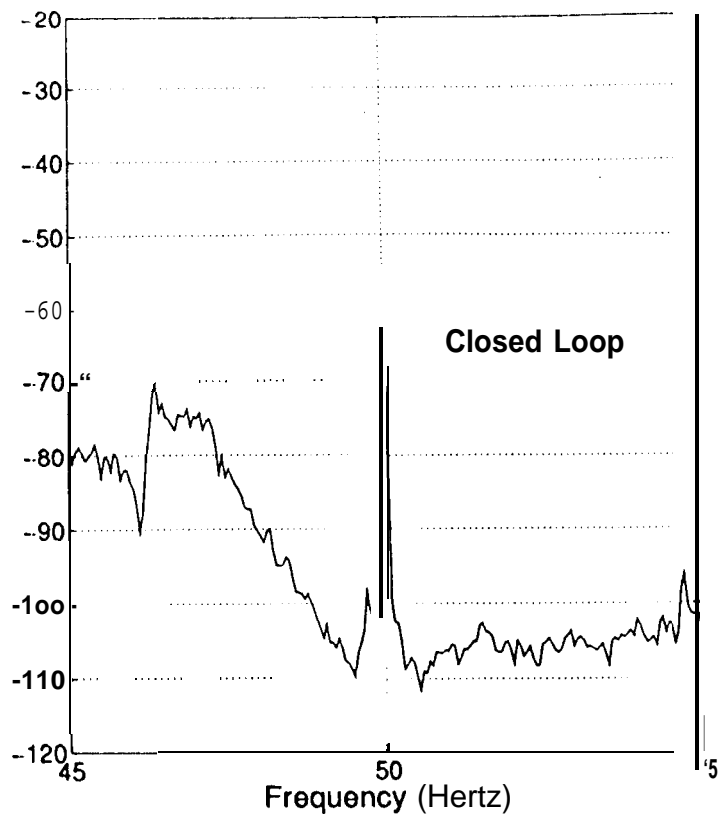
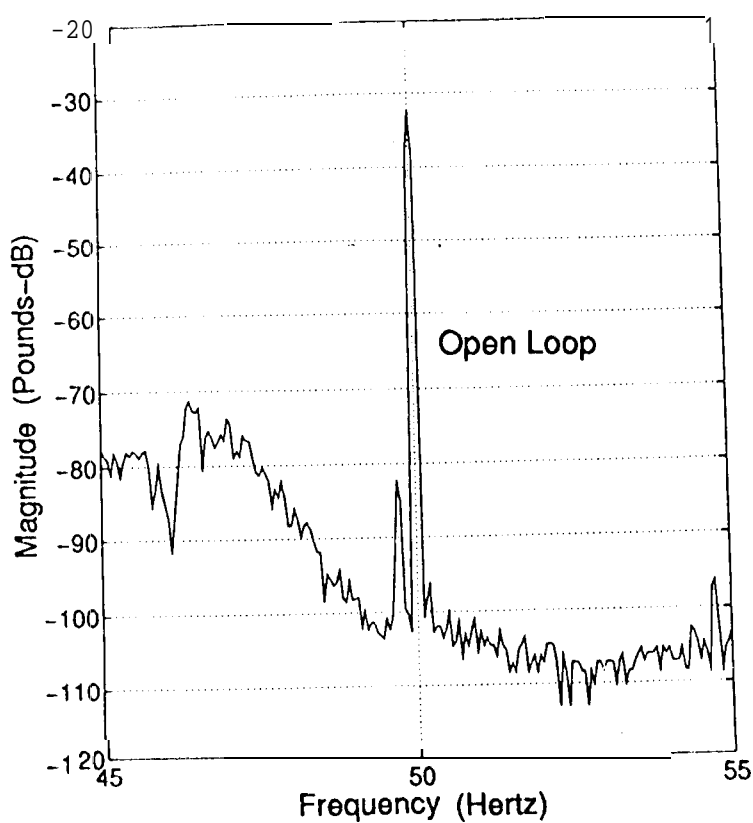


Figure 7: Isolation at 50 Hz

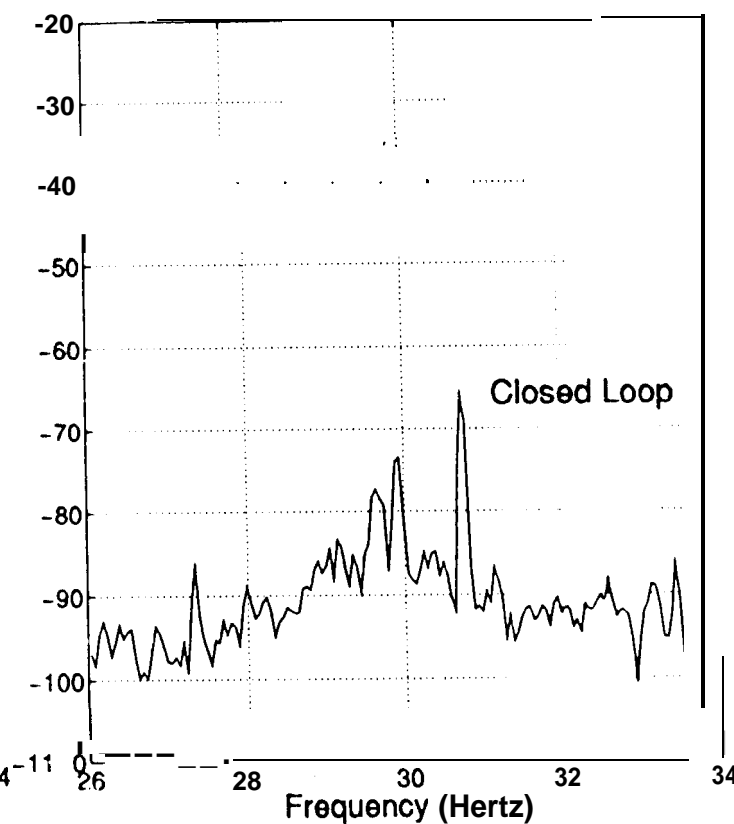
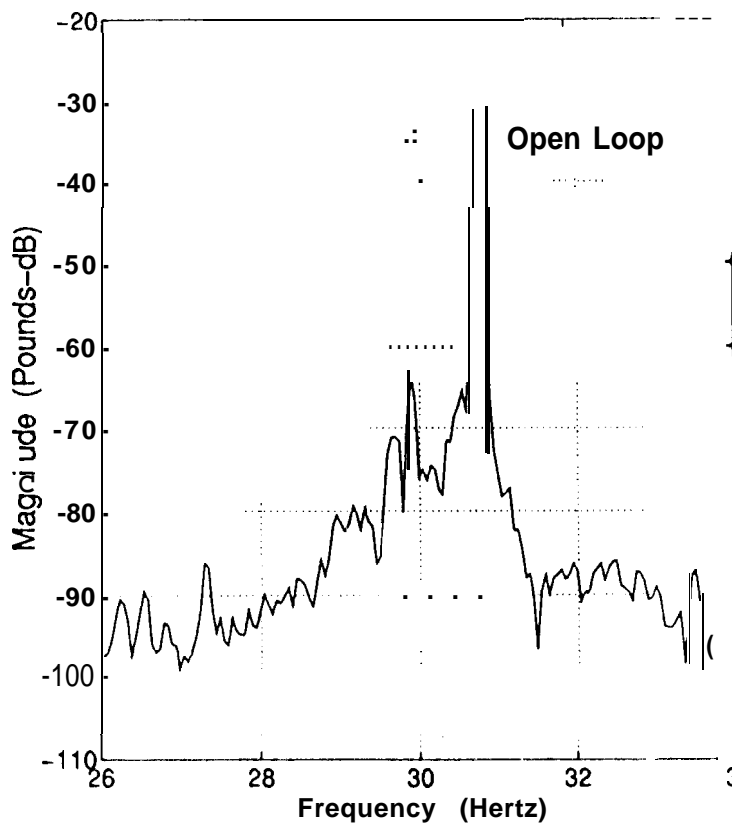


Figure 8: Isolation at resonant frequency of 30.75 Hz

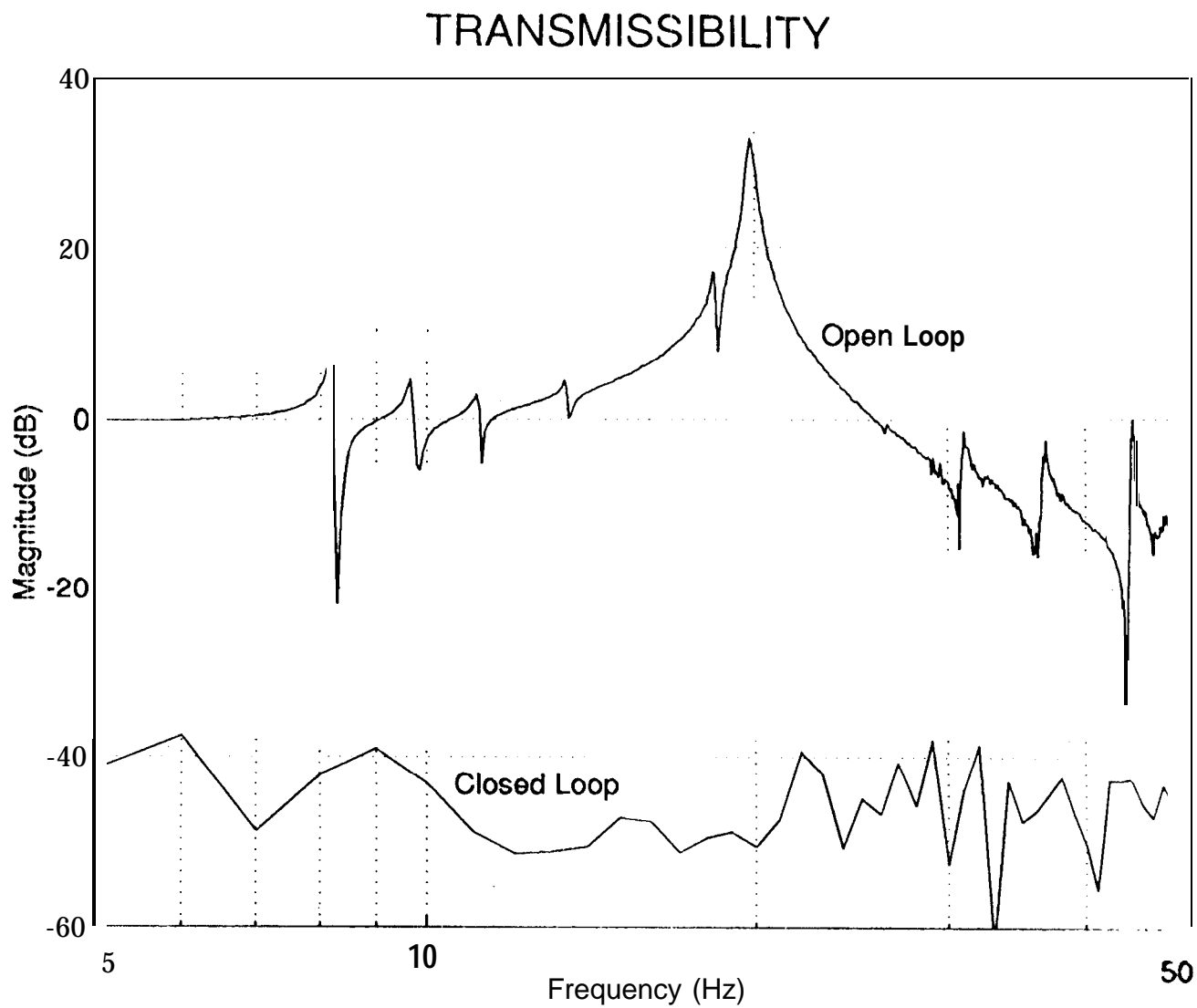


Figure 9: Disturbance transmissibility as disturbance frequency varied from 5 Hz to 50 Hz